FTD-HC-23-1837-71

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DESIGN AND TESTING OF HYDRODYNAMIC JOURNAL FRICTION BEARINGS WITH PARTIAL CAPTURE ANGLE WITHOUT RADIAL CLEARANCE

by

Ye. I. Kvitnitskiy, Yu. D. Poltavskiy

FOREIGN TECHNOLOGY DIVISION

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Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) Foreign Technology Division Air Force Systems Command U. S. Air Force	2a. REPORT SECURITY CLASSIFICATION UNCLASSIFIED
	2b. GROUP

3. REPORT TITLE

DESIGN AND TESTING OF HYDRODYNAMIC JOURNAL FRICTION BEARINGS WITH PARTIAL CAPTURE ANGLE WITHOUT RADIAL CLEARANCE

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

Translation

5. AUTHOR(S) (First name, middle initial, last name)

Kvitnitskiy, Ye. I. ; Poltavskiy, Yu. D.

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
September 1970	9	9

8a. CONTRACT OR GRANT NO.	9a. ORIGINATOR'S REPORT NUMBER(S)
b. PROJECT NO.	FTD-HC-23-1837-71
c.	9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)
d.	

10. DISTRIBUTION STATEMENT

Approved for public release; distribution unlimited.

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY
	Foreign Technology Division Wright-Patterson AFB,

13. ABSTRACT

In theoretical analyses of the operation of friction bearings without radial clearance published to date, the authors have assumed zero pressure in the minimum and maximum clearances in determining the distribution of hydrodynamic pressures. In this article, the theoretical and experimental data for bearings of this type are supplemented by considering the curvature of one of the boundaries of the load bearing area of the lubricant layer, observed with capture angles of over 60 degrees. In the calculation of bearings performed in this article, the load is considered to operate in one direction only, the surface of the bushing and journal are considered cylindrical and without skew, and the flow of the lubricating liquid is considered isothermal. Test stand experiments were performed to check the results of the calculations, showing good correspondence. [AP1018486]

DD FORM 1 NOV 65 1473

UNCLASSIFIED

Security Classification

UNCLASSIFIED
Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Hydrodynamic Bearing						
Friction						
Jcurnal Bearing						
Lubrication						

UNCLASSIFIED
Security Classification

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English pages: 9

Source: Mashinovedeniye (Mechanical Engineering) 1970, No. 5, pp. 100-106

UR/0380-70-000-005

PREPARED BY:

TRANSLATION DIVISION
FOREIGN TECHNOLOGY DIVISION
WP-AFB, OHIO.

Date 3 April 1972

UDC 621.822.5

DESIGN AND TESTING OF HYDRODYNAMIC JOURNAL FRICTION BEARINGS WITH PARTIAL CAPTURE ANGLE WITHOUT RADIAL CLEARANCE

[Article by Ye. I. Kvitnitskiy, Yu. D. Poltavskiy; Moscow, Mashinovedeniye, Russian, No 5, 1970, pp 100-106]

Friction bearings without radial clearance are used in many branches of machine building. Moreover, many bearings operating under conditions of frequent startups and shutdowns, also achieve a stated geometry of contact between the insert and shaft during the process of break-in and wear [1, 6, 7], and the new geometry alters the initial hydrodynamic characteristics of the friction bearing. The existing literature contains a few theoretical solutions of bearings of this type, done by Kingsbury [8], Pinkus [9], Tipey [6]. These authors, in determining the law of distribution of hydrodynamic pressured, used the condition of equality of the pressure to zero in the minimum and maximum clearance. It is presumed essential to supplement theoretical and experimental calculation data for this type of bearing with consideration of the curvilinearity of one of the boundaries of the bearing region of the lubricant layer, which occurs in the case of angles of capture of the journal by the insert greater than 60°.

Under the action of a load, constant with respect to direction in constructions of the examined type of bearing, as in bearings with different journal and insert radii, oil pockets are usually filled. The lubricant located within them at the moment of startup, accumulating on the shaft, penetrates into the journal between the shaft and bearing surface of the insert, with the result that the center of the insert, to which is imparted lateral displacement, occupies a position different from the center of the insert. As a certain rate of rotation is reached a wedge-shaped clearance is formed, within which forms the fluid friction regime.

In the case of action of a load that varies in terms of direction, separation of the center of the shaft from the center of the insert is

facilitated and such bearings can be made structurally as multisegmented. bearings with self-adjusting inserts[1].

For calculation of bearings under the action of a load constant with respect to direction, we will make the following assumptions: 1) the surfaces of the journal and insert are cylindrical and are adjusted without mutual play; 2) the flow of the lubricating liquid is isothermic.

For the calculation we will introduce the following symbols: r is the radius of the journal and insert with consideration of capture Ω; e is eccentricity; h is variable thickness of the lubricating film in the direction of flowing angle ϕ; P is load; p is pressure in the lubricating film; ϕ_0 is the angle of the load; y is the direction of length of the insert l; $\lambda = l/2r = l/d$ is the relative length of the bearing; μ is the viscosity coefficient of the lubricant.

The arbitrary working position of the journal and insert is depicted in Figure 1.

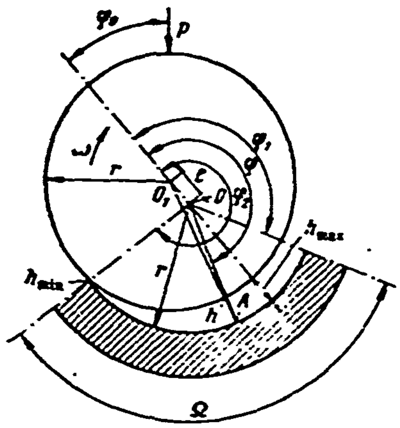


Figure 1. Position of journal and insert.

Neglecting the value $LOAO_1$, we will determine the clearance h from the equality

$$h = e\cos\varphi. \tag{1}$$

As the journal rotates with angular velocity ω the distribution of pressures in the clearance is described by the Reynolds equation

[1]An example of such constructions are guide bearings of vertical hydroturbine shafts manufactured by KhTGZ [Khar'kovskiy turbogeneratornyy zavod im. S. M. Kirova; Khar'kov Turbogenerator Plant im. S. M. Kirov]; certain types of journal bearings of metal cutting mills of the "Fil'matik" type of the Cincinnati firm, among others, are also examples of this type.

$$\frac{1}{r^2}\frac{\partial}{\partial\varphi}\left(h^3\frac{\partial p}{\partial\varphi}\right)+\frac{\partial}{\partial y}\left(h^3+\frac{\partial p}{\partial y}\right)=6\mu\omega\frac{dh}{d\varphi}. \qquad (2)$$

Converting to the new variables: $\bar{y} = y/r$, $\bar{h} = h/e$, $\bar{p} = pe^2/\mu\omega r^2$, we represent equation (2) in dimensionless form

$$\frac{\partial}{\partial\varphi}\left(\bar{h}^3\frac{\partial\bar{p}}{\partial\varphi}\right)+\frac{\partial}{\partial\bar{y}}\left(\bar{h}^3\frac{\partial\bar{p}}{\partial\bar{y}}\right)=6\frac{d\bar{h}}{d\varphi}. \qquad (3)$$

We will replace the continuous field of integration with rectangular region R with net pitch m and n and write the finite-difference form of equation (3)

$$\bar{p}_{ik}=a_{ik}\bar{p}_{i-1,k}+b_{ik}\bar{p}_{i+1,k}+c_{ik}\bar{p}_{i,k+1}+d_{ik}\bar{p}_{i,k-1}+f_{ik}, \qquad (4)$$

where

$$a_{ik}=\frac{m^2(3n\sin\varphi_{ik}+2\cos\varphi_{ik})}{4\cos\varphi_{ik}(m^2+n^2)},\qquad b_{ik}=\frac{m^2(2\cos\varphi_{ik}-3n\sin\varphi_{ik})}{4\cos\varphi_{ik}(m^2+n^2)},$$

$$c_{ik}=d_{ik}=\frac{n^2}{2(m^2+n^2)},\qquad f_{ik}=\frac{3\sin\varphi_{ik}m^2n^2}{\cos^3\varphi_{ik}(m^2+n^2)}.$$

The boundary conditions are written in the form

$$\bar{p}|_{\varphi=\varphi_1}=\bar{p}|_{\varphi=\varphi_2}=0, \qquad (5)$$

$$\bar{p}|_{\bar{y}=\pm\lambda}=0, \qquad (6)$$

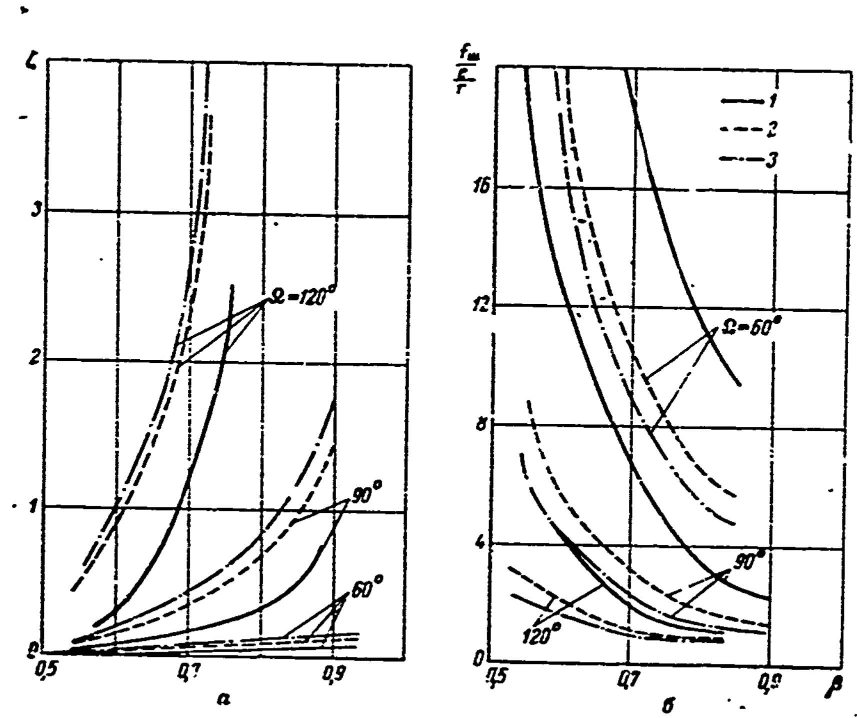
$$\bar{p}|_{\varphi=\varphi_k(\bar{y})}=\left.\frac{\partial\bar{p}}{\partial\varphi}\right|_{\varphi=\varphi_k(\bar{y})}=0. \qquad (7)$$

Condition (7) indicates curvilinearity of the boundary of formation of the bearing region of the lubricant film in the direction of length of the bearing and absence of a zone of negative pressures in the range of the solution.

Solving the boundary problems (4)-(7) by one of the numerical methods[1], we obtain the required working characteristics of the bearing. Thus the load factor is expressed in the form

$$\zeta=\frac{P(e/r)^2}{ld\mu\omega},$$

[1]The method proposed by Buleyev [2] is used in this work.



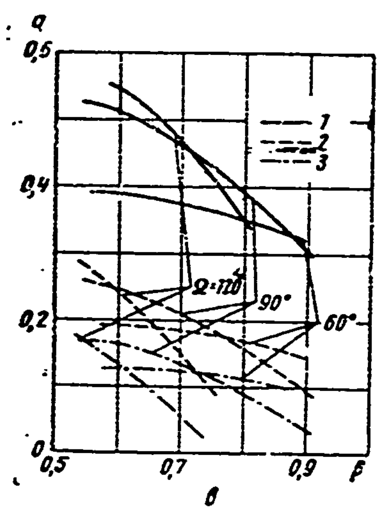


Figure 2. Dimensionless characteristics of bearing as function of β for various capture angles Ω and relative length λ: 1 -- $l/d = 0.5$; 2 -- $l/d = 1.0$; 3 -- $l/d = 1.5$.

For the purpose of checking the calculation data obtained we made an experimental study on a stand [5], the test specimen for which was a hydrodynamic bearing with the dimensions $l = 270$ mm, $d = 250$ mm and $\Omega = 120°$.

The directions of the action of the load were changed and fixed relative to the ends of the insert.

The load on the insert varied from 120 to 5,000 kg, the rpm from 600 to 1,500. Grade 22 turbine oil was used as the lubricant.

During the tests we used: thickness of lubricating film at ends of insert, determining the mutual position of the centers of the shaft and insert, distribution of pressures in the lubricating film, temperature of the lubricating film at bearing inlet and outlet, shaft rpm.

For measuring the thickness of the lubricating film we used four small-scale inductive sensors with a П-shaped core [3]. The placement of the sensors made it possible to monitor the thickness of the film both on an arc and on the generatrix of the insert bearing surface.

Fittings with an axial channel (1.5 mm in diameter), connected to a cylindrical sheath, to the wall of which was attached the working resistance strain gauge, were inserted for measurement of the hydrodynamic pressures at six individual points of the insert. Deformation of the sheath walls under the influence of pressure, transmitted through the mass of the hydraulic layer in its cavity and axial channel, caused the measurement bridge of the circuit to go out of balance. The hydraulic layer was used as an incompressible substance capable of transmitting a signal of the measured pressure without distortion. The arrangement of the sensors made it possible to measure pressure on the arc of the insert at its midpoint and along its length in one of the cross sections.

The above-described sensors operated in combination with the standard apparatus: 8ANCh-7M amplifiers and N-102 loop oscillographs.

Furthermore, we measured continuous diagram of pressures, which enabled us to analyze qualitatively and quantitatively the hydrodynamic process of lubrication, determine the boundaries of the bearing region of the oil film. For this purpose, in contrast to the known measurement schemes using contact and mercury terminals, we developed a procedure for measuring pressures based on the use of a capacitance sensor and contactless terminal device of the induction type [4].

The design of the capacitance pressure sensor, installed in a radial boring of the shaft, is simple and represents a capacitance formed by a mutually insulated steel membrane (active diameter of 8 mm, 0.1 mm thick) and electrode.

The space between the membrane and the electrode is controlled and selected as a function of the measured pressure. The pressure that deforms the membrane alters the capacitance of the sensor, which is recorded by the measurement instrument. In it the measured capacitance enters the contour of the master high-frequency generator, operating under conditions of resonance or close to it. As the capacitance changes, the resonance breaks down, which leads to a change of the other characteristics of the circuit. For transmission of the signal from the master generator a contactless induction terminal, representing a transformer, one of whose windings is rotated along with the sensor and makes with it a parallel resonance contour, and the other of which is connected to the stationary measurement apparatus, was used for transmitting the signal from the

where

$$P = \sqrt{P_1^2 + P_2^2},$$

$$P_1 = P\cos\varphi_0 = r\int_{-l/2}^{+l/2}\int_{\varphi_1(y)}^{\varphi_2} p\cos\varphi\, d\varphi\, dy, \quad P_2 = P\sin\varphi_0 = r\int_{-l/2}^{+l/2}\int_{\varphi_1(y)}^{\varphi_2} p\sin\varphi\, d\varphi\, dy.$$

Here $\phi_2 = \beta\Omega + \pi$, where β determines the position of center line OO_1. Thus

$$\zeta = \zeta(\Omega, \lambda, \beta).$$

The load angle ϕ_0 at which the system is in equilibrium is determined from the expression

$$\varphi_0 = \operatorname{arctg}\frac{P_2}{P_1}.$$

The force of friction on the journal and bearing is

$$F_{\tau_{j,b}} = \mu\omega \bar{d} l \frac{r}{c}\xi_{\tau_{j,b}},$$

where

$$\xi_{\tau_{j,b}} = \int_{-\bar{\nu}(\lambda)}^{+\bar{\nu}(\lambda)}\int_{\varphi_1(\bar{y})}^{\varphi_2}\left(\frac{1}{\cos\varphi} \pm \frac{\cos\varphi}{2}\frac{\partial\bar{p}}{\partial\varphi}\right)d\varphi\, d\bar{y}$$

is the coefficient of resistance to rotation.

We will express losses to friction as the dimensionless function $f_m/(e/r) = \xi_{fj}/\zeta$, where f_j is the coefficient of friction on the journal.

The amount of lubricant flowing through the bearing is

$$Q = \int \frac{h^3}{12\mu}\frac{\partial p}{\partial y}\bigg|_{y=\pm\frac{l}{2}} r\, d\varphi = q\omega l d e,$$

where q is the dimensionless coefficient characterizing the lubricant flow rate.

The characteristics ζ (Figure 2,a), $f_j/(e/r)$ (Figure 2,b), and q (Figure 2,c) are represented in Figure 2 as functions of β for various typical dimensions, determined as a result of calculations on the M-20 electronic digital computer.

master generator. Between the windings is a strong inductive connection, thanks to the equal length of their winding and concentric arrangement with a diametral clearance of 1.5 mm.

The sensors were calibrated directly in their operating position.

The temperature of the lubricating film was measured with the aid of chromel-kopel thermocouples, attached to the entrance and exit edges of the bearing. At the points of build-up of the surrounding metal the thermocouples were thermally insulated with glass fiber permeated with phenolformaldehyde resin. The emf of the thermocouples was measured with a KP-59 potentiometer.

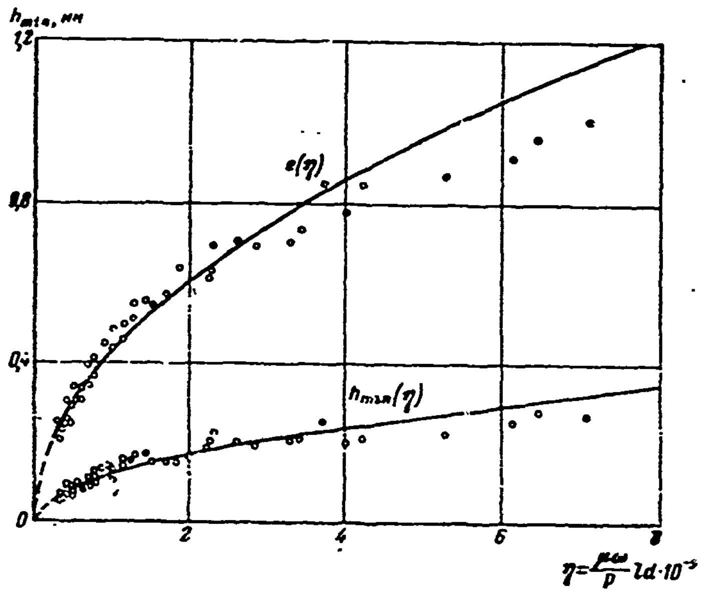


Figure 3. Value of e and h_{min} as function of η with direction of load lagging behind entrance edge of insert by 89°.

The rpm was measured with an induction sensor of the design described above. The sensor was fixed relative to the ends of the bearing.

As a result of the tests we obtained a complex interrelation between the investigated parameters, which reduces to the dependence of h_{min} and e on the dimensionless characteristic of the mode of operation

$$\eta = \frac{\mu\omega}{p} ld.$$

The theoretical curves and experimental data plotted in the form of individual points for one of the fixed load directions are shown in Figure 3. The dependences that are represented, both theoretical and

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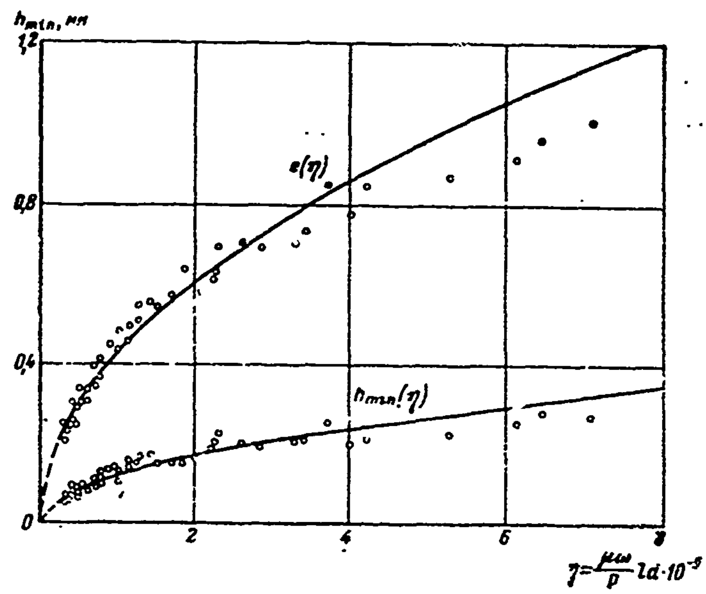

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The theoretical curves and experimental data plotted in the form of individual points for one of the fixed load directions are shown in Figure 3. The dependences that are represented, both theoretical and

experimental, are of parabolic character, and in the lightly loaded regime the discrepancies between them are maximal, reaching up to 25%, which is explained by possible deviations of the process from stationarity and general error of the experiment. The stated discrepancies did not exceed 12% in the other loading regimes.

Additional illustration of the comparison of the theoretical and experimental data is seen in Figure 4, where the dimensionless functions $P(\phi)$ are presented for the same load direction, as well as the mutual location of the lines of centers and loads. In this case the operating mode corresponded to $\eta = 0.8$ ($P = 2{,}385$ kg, $\omega = 83.73\ \text{sec}^{-1}$, $\mu = 2.37 \times 10^{-3}$ kg•sec/m^2). In Figure 4, 1 are theoretical data; 2 are experimental data, and the curve $P(\phi)$ is plotted with the aid of the capacitance sensor. The points here represent the results established by individual pressure sensors. The origin of the bearing region of the lubricating film for all loading angles was located in the diffuser section of the clearance, i.e., prior to joining, where $h = h_{max}$, which indicates the validity of boundary condition (7).

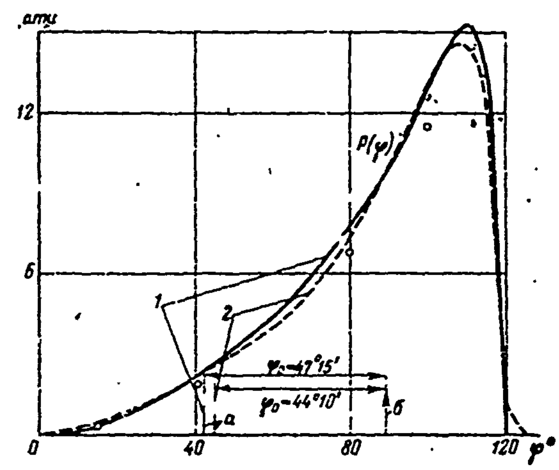

Figure 4. Distribution of pressures around circumference of insert and mutual distribution of lines of centers (a) and load (b).

The presence of an oil pocket in the exit end of the bearing has an effect on the breakaway of the bearing region of the lubricating film, which, judging by the pressure diagram, even though pronounced, reaches values $P = 0$ beyond the edge, where $h = h_{min}$.

This fact is explained by the small angle of diffusivity of the oil pocket cross section beyond the edge of the minimal clearance, with the result that the clearances, symmetrically taken relative to the stated edge on a 5-7° segment of the arc on both sides of the shaft rotation, are commensurate.

As a whole the theoretical dependences presented above revealed sufficient agreement with experimental research data, which permits them to be used in engineering practice.

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